

Multibody dynamic
modelling racing car
suspension biology
essay



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There is no one perfect suspension design, optimising any suspension system is an iterative process of constant trade-offs and compromise involving continuous trial and error methodology. In order to enhance performance in one area, it will nearly always mean sacrificing performance in another. For example, a shorter wheelbase improves cornering response, but makes for a less stable ride. Balancing these trade-offs, by tweaking the design and continuously analysing the feedback, is fundamental to successful suspension design.

It would obviously be costly, in terms of finance, man hours, materials and waste, to produce and subsequently examine prototypes for the essentially infinite number of suspension configurations that could be investigated. Instead, MSC Adams, a specialised software package for multibody dynamic simulation will be used to create a CAD model of the suspension and then to measure a multitude of parameters under defined conditions. Modifications can be made to the design of the suspension design very quickly and the results can be compared visually.

1. 1. 2 The Process

In order to progress with the suspension design, the fundamental parameters must be determined. The FSAE rulebook defines certain limitations on the dimensions of the suspension assembly and its various components. UCD's previous entry car is also at hand as a basis for further design.

Every car in the 2012 event utilised some form of double unequal length A-arm suspension (double wishbone). This design allows critical parameters such as camber, caster and toe angles; roll centre height; scrub radius and

scruff to be controlled throughout suspension travel. Dimensional design data including length, width, height and wheelbase; the track width, front and back; the car's weight and its distribution; the wheels and tyres used; fuel system used and maximum power/torque output is available for all of the previous year's entries.

Having used information from the rules, dimensional and analytical data from UCD's own previous entry and the records of previous year's entries to develop a basic starting-point model, it is then a matter for the engineering team to determine how they want the car to handle. From there the iterative process begins to engineer the required performance by tweaking the various parameters in the CAD model using MSC Adams. The effects of these changes must be thoroughly analysed and compared with previous configuration plots.

1. 2 Software

MSC Adams is the most used multibody dynamics and motion analysis software in the world. Such software is preferred to Finite Element Analysis alternatives because it incorporates real physics, simultaneously solving equations for kinematics, statics, quasi-statics and dynamics, providing solutions in a fraction of the time required by FEA (MSC Software, 2013).

Licensing this software is extremely expensive but fortunately MSC run a Formula Student sponsorship program, making five licences available for free in exchange for placement of their logo on the car's body.

Several subprograms exist within Adams for different simulation applications; one of these subprograms, Adams/Car, is tailored to vehicle simulations and so will be the most important of these.

Building a Formula Student racing car model from scratch would be very difficult and immensely time-consuming for novices such as students just getting to grips with the enormous complexity of the software. Fortunately, over the years an open-source, fully customisable FS template was created by MSC employees and made freely available to license holders. This offers a great base from which to begin optimising the suspension.

Every dimension, parameter and material characteristic can be defined in the model and the geometry of the suspension can be adjusted by adjusting the hardpoints of the model. These are essentially the Cartesian coordinates of key components.

The software can be used to plot almost any combination of parameters against each other under a multitude of customisable conditions. The effects of configuration modifications can be assessed by comparing these results. All simulations are written to files and can be visualised in customisable graphs for comparative studies.

In Adams/Car, the suspension subsystems can be modelled independently and then assembled into full vehicle. This allows very quick changes of entire subsystems such as front and rear suspensions, steering rigs, front and rear wheels, and the chassis. If further detail is required, further subsystems such as brakes, anti-roll bars and powertrain can be implemented. The

subsystems interact via communicators; there are input and output communicators that read and send information respectively.

The suspension modelled here with then be implementable into a full vehicle and track simulation using the same software package. A feature called SmartDriver uses an inbuilt quasi-static solver to calculate a speed profile along a defined path, allowing simulation of the maximum or target performance as defined by the operator. This feature allows one to readily compare handling, durability and ride performance in various design configurations over a specific course. (MSC Software, 2009).

2 Literature Review

2.1 Formula Student

2.1.1 History

Backed by companies like Airbus, Jaguar Land Rover, National Instruments and Shell along with Mercedes AMG Petronas Formula One Team Principal Ross Brawn, Formula Student is the leading educational motorsport competition in Europe. The program is run in partnership with the Society of Automotive Engineers (SAE) in the United States who started Formula SAE in 1981 (Formula Student, 2013).

There are two entry categories for fully constructed vehicles (1 and 1A) and two corresponding categories for entries without a complete car (2 and 2A), based on design, presentation and cost. Class 1A builds on Class 1 to encourage sustainability, requiring that the design incorporates novel powertrain technologies. Dynamic events are the same but the cost category

is replaced by sustainability, which is marked on CO2 emissions and embodied energy in the vehicle.

3000 students, making up 132 teams, from 34 countries participated in the 2012 event as the competition continues to grow year-on-year. 43 of these teams entered using alternative fuels, 33 in class 1A and ten in class 2A.

Fuels used in 2012 included E85 (85% Ethanol + 15% Gasoline), Electric and Hydrogen (Formula Student, 2012).

2. 1. 2 Formula Student in UCD

The UCD team plan to enter their latest car in the 2015 Class 1A event. The 1A category was chosen to encourage progressive thinking in terms of sustainability and with a view to attracting sponsorship from enterprises with a vested interest in the development of alternative automotive fuels.

2. 1. 3 Competition

The prototype car produced should be low cost, easily maintainable and reliable. They must also perform well in acceleration, braking and handling. Scoring is based on four static events: cost and presentation; technical and safety scrutineering; tilt test; and brake and noise test, and four dynamic events: skid pad; sprint; acceleration; and endurance and fuel economy (Formula Student, 2013).

2. 1. 4 Rules

SAE International outline the technical regulations for participation in the Formula Student competition. These rules cover the administrative requirements and all elements of the technical design. There are a number

of regulations referring specifically to the suspension and more with indirect relations. Article 6 of the regulations addresses the general chassis rules and section 6. 1 defines fundamental suspension limitations (SAE International, 2012):

T6. 1. 1: The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50. 8 mm (2 inches), 25. 4 mm (1 inch) jounce and 25. 4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate handling inappropriate for an autocross circuit.

T6. 1. 2: All suspension mounting points must be visible at Technical Inspection, either by direct view or by removing any covers.

T2. 3: The car must have a wheelbase of at least 1525 mm (60 inches). The wheelbase is measured from the centre of ground contact of the front and rear tires with the wheels pointed straight ahead.

T2. 4: The smaller track of the vehicle (front or rear) must be no less than 75% of the larger track.

T6. 3. 1: The wheels of the car must be 203. 2 mm (8. 0 inches) or more in diameter.

The limitations placed on suspension design are evidently kept to a minimum. Accordingly, there is great variation in suspension system design between teams.

2. 2 Vehicle Axis System

Figure 2. 2: vehicle axis system (Milliken, W. F. & Milliken, D. L., 1995).

The axis system described in the above figure were defined by the Society of Automobile Engineers in 1976 to ensure consistency in industry publications (Carroll, W., 2002). The x-axis points forward in the horizontal plane, the y-axis to the drivers right in the same plane and the z-axis points downward. The origin lies at the centre of gravity of the system.

2. 3 Transient Stability and Control

2. 3. 1 Spring-Mass-Damper System

Figure 2. 3. 1: spring-mass-damper system (Milliken, W. F. & Milliken, D. L., 1995).

There are numerous applications in which the dynamics of rigid bodies are approximated by a system like the one shown in figure 2. 3. 1, including wheel suspension systems. Friction present in actual suspension systems is neglected. The weight component can also be eliminated by taking a zero reference at rest.

The dynamics of this SMD system are completely defined by two parameters: its undamped natural frequency and its damping ratio.

2. 3. 1. 1 Undamped Natural Frequency

The undamped natural frequency is the frequency at which a mass will oscillate, about the zero reference, if the mass is displaced and released. It can be calculated by

, Eq. 2. 3. 1. 1

where is the spring constant and the mass. In theory, if the spring is assumed to dissipate no energy, the mass would continue to oscillate indefinitely.

2. 3. 1. 2 Damping Ratio

The damping ratio is a mathematical tool to quantify the influence of damping constant, c , on transient response. The damping ratio is calculated by

. Eq. 2. 3. 1. 2

Critical damping occurs at where the mass will return to zero reference without under/overshoot. A system is underdamped if and the mass will oscillate about the zero reference with decreasing amplitude until it eventually reaches steady-state. In an over damped system, where $\zeta > 1$, the mass will return smoothly to zero reference but slower than it would at critical condition.

2. 3. 2 Quarter-Car Model

Figure 2. 3. 2: quarter-car spring-mass-damper system (Milliken, W. F. & Milliken, D. L., 1995).

2. 3. 2. 1 Chassis Natural Frequency

To establish the chassis natural frequency, the suspension spring rate, k_s , must be determined. This is achieved by measuring ride rate, R , knowing the tyre spring rate, k_t , and using the equation

. Eq. 2. 3. 2. 1. 1

The natural frequency can then be calculated by

. Eq. 2. 3. 2. 1. 2

2. 3. 2. 2 Wheel Natural Frequency

The wheel natural frequency, or wheel hop frequency, is calculated using

. Eq. 2. 3. 2. 2

2. 3. 2. 3 Shock Absorber

To calculate the required damping constant for the sprung mass, first, a suitable damping ratio must be chosen based on preference for over/underdamping. The following may then be used to calculate :

. Eq. 2. 3. 2. 3. 1

For the unsprung mass, in order to keep the chassis under control, wheel motion should be slightly overdamped, and so it is important to determine its resulting damping ratio. This can be calculated using

. Eq. 2. 3. 2. 3. 2

2. 4 Springs and Dampers

Harris (2006) details the factors which must be considered in the design of the suspension with regard to spring and damper configurations. The springs permit the relative motion between the wheels and chassis while dampers suppress oscillations in the motion. The spring rate, the ratio of applied force to deflection of spring, is of primary concern as they determine how the car

will manoeuvre; if too low, there will be too much ride and roll, if too high, the wheels will not maintain contact with the road.

When a spring is deformed, potential energy is stored in the spring; when the spring is free to undeform, the spring would, in the absence of energy loss to heat or friction, oscillate indefinitely. Damping is used to reduce, or eliminate, these oscillations. Implementing too much damping will degrade ride comfort, but the car will handle very well and vice versa. Once again, a compromise must be sought, but in racing the car's handling will be prioritised over ride comfort.

There is no one perfect damping ratio, conditions change from track to track and different drivers prefer different setups. It is best to begin with critical damping and iterate through slight under and overdamping scenarios. Harris (2006) suggests a slightly underdamped (0.5-0.7) setting may be optimal. Furthermore, the author advises that damping coefficient in bump should be approximately half the value in rebound.

2.5 Suspension Kinematics

The properties of a suspension important to the dynamics of the vehicle are primarily seen in the kinematic (motion) behaviour and its response to the forces and moments that it must transmit from the tyres to the chassis (Gillespie, 1992). The primary role of the suspension system in racing car design is to keep the tyres in constant contact with ground; this allows the driver to maintain traction to the limit of their capacity (Theander, A., 2004).

The double a-arm suspension geometry is the preferred option for racing car designers in all forms of racing (Carroll, W., 2002). This design has the <https://assignbuster.com/multibody-dynamic-modelling-racing-car-suspension-biology-essay/>

advantages of being able to accurately control the kinematics of the wheel and that it is relatively easy to calculate the loads in various parts of the system. It is also a relatively lightweight option.

2. 5. 1 Suspension Geometry

The geometry of the suspension determines the motion of the wheel and the rate at which certain characteristics of the suspension change due to this motion. These characteristics include camber angle, toe angle, dive, squat and lift (Harris, C., 2006).

While the properties of the various suspension components impact on the performance of the system, the geometry of their configuration determines how the loads are distributed and transferred under various accelerations. The layout also determines the motion of each wheel, and subsequently the tyre area in contact with the ground, under conditions of acceleration, braking and cornering.

2. 5. 1. 1 Wheelbase and Track Width

The wheelbase is the horizontal distance between the centres of the front and back axles of the vehicle. This dimension has a very important role in load distribution between front and back and load transfer under longitudinal accelerations. A longer wheelbase reduces the load transfer backward and forward under acceleration and deceleration respectively; this allows for softer springs to be used, improving ride comfort. However, a smaller wheelbase makes for a more agile car with better handling and a smaller turning radius.

The front and rear track widths are the distances between centres of the front and rear tyres respectively. This dimension is important in cornering as it resists the overturning moment due to the inertia force at the centre of gravity, the lateral forces produced in the tyre contact patch and the lateral load transfer in cornering. A narrower track is preferred for tight, twisty circuits like those seen in Formula Student. Gaffney and Salinas (1997) explain that in order to increase rear traction during corner exit, it helps to have less roll resistance in the rear than at the front. To achieve this, the front track width will typically be greater than at the rear for rear wheel drive vehicles. This layout also has the added benefit of allowing the driver to position the front wheel as close to the apex as possible without worrying about how the rear wheel will follow.

2. 5. 2 Suspension Terminology

2. 5. 2. 1 Camber

Figure 2. 5. 2. 1: camber angle (Milliken, W. F. & Milliken, D. L., 1995).

Camber is the lateral inclination of the wheel with respect to the vertical plane as in figure 2. 5. 2. 1. The camber angle ($\dot{\theta}$) is considered negative when the top of the wheel leans towards the centre of the vehicle. The change in camber angle per unit displacement of the wheel centre relative to the sprung mass is defined as the rate of camber change (Carroll, W., 2002).

Maximum cornering force is achieved at a small negative camber angle, however, if the magnitude is too great then the effective width of the tyre is reduced (Harris, C., 2006) and consequently its ability to produce frictional

forces is greatly diminished (Carroll, W., 2002). It is therefore important to minimise the rate of camber change.

A cambered wheel produces a lateral force in the direction of the tilt; at zero slip angle this is known as camber thrust (Theander, A., 2004).

2. 5. 2. 2 Kingpin Inclination, Spindle Length, Scrub Radius

Figure 2. 5. 2. 2: front view kingpin geometry (Milliken, W. F. & Milliken, D. L., 1995).

The kingpin inclination is the angle between the vertical plane and a line projected through the upper and lower ball joints in the y-z plane, as shown in figure 2. 5. 2. 2. It represents the lateral tilt of the axis around which the wheel rotates when it is steered. In contrast to camber angle, the kingpin inclination is defined as being positive when the axis is tilted inboard (Carroll, W., 2006).

For any non-zero kingpin inclination the car will be raised when the front wheels are steered to either side; this effect is amplified with increasing inclination magnitude. With positive inclination the front wheels will tend to camber positively in steering (Milliken, W. F. & Milliken, D. L., 1995). The effect is small but may be significant in a tight circuit as in Formula Student.

The spindle length, as demonstrated in figure 2. 5. 2. 2, is, for the sake of packaging, generally found to be positive. Increasing the spindle length also increases the lifting effect associated with steer (Carroll, W., 2002).

Scrub radius, the distance between the intersection of the ground and the steering axis and the vertical plane through the wheel centre, is said to be negative when this intersection point is outboard of the wheel centreline as in figure 2. 5. 2. 2. Carroll (2002) suggests a small amount of positive scrub is preferred to provide the driver with steering feedback. If the accelerating or braking force is asymmetrical, a steering torque proportional to the scrub radius will be felt at the steering wheel (Theander, A., 2004).

2. 5. 2. 3 Caster, Mechanical Trail

Figure 2. 5. 2. 3: side view kingpin geometry (Milliken, W. F. & Milliken, D. L., 1995).

Caster angle, shown in positive arrangement in figure 2. 5. 2. 3, is the angle the kingpin (steering) axis makes with the vertical in the x-z plane. Like kingpin inclination, the caster angle causes the sprung mass to be raised with steer, however, with caster angle the effect is not symmetrical. With equal positive camber the weight will be shifted diagonally in cornering; in a left-hand turn the load will be increased on the left-front and right-rear tyres. This effect leads to an oversteering effect (Milliken, W. F. & Milliken, D. L., 1995).

Caster angle also influences steer-camber, however, unlike kingpin inclination, the steering leads to an increase in negative camber in the outside wheel and to an increase positively in the inner wheel; subsequently, both tyres are leaning into the turn. As Carroll (2002) points out, this is a preferred outcome as, due to lateral load transfer, the majority of the forces will be borne in the outside wheel.

The mechanical trail, or caster offset, shown as positive in figure 2. 5. 2. 3, is analogous to scrub radius but in the side profile; it is the distance between the intersection of the steering axis and the ground and the centre of the tyre print. Theander (2004) indicates that the steering torque generated is proportional this trail.

2. 5. 2. 4 Wheel Toe

Toe-in describes the condition whereby the wheels, when viewed from above, are pointed inwards; conversely, for toe-out the wheels point outwards. Clancy (2006) outlines how it is possible to control the toe angle through the wheels vertical motion. The geometry can be arranged to give either toe-in or toe-out in bump or rebound.

According to Theander (2004), minimum static toe is preferred to reduce roll resistance and tyre heating and wear, however, rear toe can be used to help with turn-in and front toe can be configured based on preference for Ackermann steering, ride and roll steer, compliance steer and camber.

2. 5. 2. 5 Ackermann Steering

Figure 2. 4. 2. 5: Ackermann steering, parallel steer and reverse Ackermann (Theander, A., 2004).

In cornering, the curve radius followed by the inner wheel must be smaller than the outer wheel. It is possible to configure steering linkages to achieve any of the three scenarios described in figure 2. 5. 2. 5. Clancy (2006) indicates that this decision should be made based on the tyre characteristics.

By plotting lateral force as a function of slip angle at various vertical loadings, it is possible to determine the optimum configuration.

2. 5. 2. 6 Instant Centre, Roll Centre, Jacking

Figure 2. 5. 2. 6: instant centre, roll centre (Theander, A., 2004).

The instant centre is an imaginary point about which the suspension linkage instantaneously pivots. Figure 2. 5. 2. 6 shows the projected instant centre for a double a-arm suspension geometry, this point moves with any changes in this geometry. The front view swing arm (fvsa) represents a geometrically equivalent single suspension link, its length is also shown in figure 2. 5. 2. 6. According to Carroll (2002), variation - position and rate of change - in the instant centre is directly determined by the suspension geometry.

In order to determine the roll centre of the sprung mass of the car, a line is drawn in the front elevation from the centre of the contact patch to the instant centre for both sides of the car; the intersection of these lines denotes the roll centre. Theander (2004) illustrates how the roll centre establishes the coupling point between the sprung and unsprung masses of the car. In cornering, the centrifugal force acting on the centre of gravity is translated, through the roll centre, to the tyres where reactive lateral forces are developed.

Carroll (2002) goes on to describe an effect known as jacking. If the roll centre lies above ground level, the lateral force will create a moment pushing the wheel down and thus raising the sprung mass and vice versa for the roll centre lying below ground level. A compromise in roll centre height must be sought in order to ensure minimal body roll and jacking.

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2. 5. 2. 7 Anti Features

Figure 2. 5. 2. 7: braking anti features (Theander, A., 2004).

Milliken and Milliken (1995) state that forces in the longitudinal direction are controlled by the side view swing arm (shown in figure 2. 5. 2. 7). The effects of this geometry are borne in the anti-dive, anti-lift and anti-squat parameters; these 'anti' parameters describe the longitudinal to vertical force coupling between sprung and unsprung masses. The side view instant centre is typically behind and above the front wheel centre and ahead and above most rear suspensions.

Anti-dive in front suspensions reduces bump deflection under braking; anti-lift reduces droop travel in forward braking; and anti-squat reduces bump travel during acceleration. In order to minimise complexity, these anti-effects were not incorporated in the UCD-R4 car's suspension geometry (Clancy, S., 2006).

2. 6 Tyre Performance

Figure 2. 6: SAE tyre axis system (Milliken, W. F. & Milliken, D. L., 1995).

The SAE tyre axis system shown in figure 2. 6 has its origin at the centre of the stationary tyre footprint and assumes a flat and level road. The diagram also denotes a positive slip angle (described in section 2. 6. 2. 1). Inclination angle differs from camber angle only in sign convention. Profiling a negatively cambered from the rear, both wheels would be said to have negative camber but only the right wheel has negative inclination angle, the left wheel has positive inclination as in figure 2. 6.

2. 6. 1 Tyre Contact Patch

The complex interaction of the tyre and road combination at the contact patch provides the tractive, braking and cornering forces required to manoeuvre the vehicle (Clancy, S., 2006) and as stated by Gillespie (1992), it has been said that “ the critical control forces that determine how a vehicle turns, brakes and accelerates are developed in four contact patches no bigger than a man’s hand”.

Carroll (2002) asserts that the forces generated in the tyre contact patch are dependent on sliding velocity, local load and the characteristics of both the road and the tyre rubber. This mechanism is not well understood, but what is more important is that the forces can be determined in the print and those forces can be apportioned to two types of friction, hysteresis due to slip and adhesion.

2. 6. 2 Tyre Forces

2. 6. 2. 1 Lateral Force, Slip

Tyre Basics: Shows changes in the contact patch with no slip angle, initial slip angle and mature slip angle.

Figure 2. 6. 2. 1: slip angle (Creative Car Control, 2013).

By SAE standards, lateral forces originate at the centre of the tyre contact patch and propagate perpendicular to the direction of the wheel if no inclination or camber is present. These forces are critical as they provide the bulk of the force required to change the vehicle heading. According to Carroll

(2002), it is often this lateral force that limits the performance of a racing car.

If a lateral force, acting through the hub centre, is applied to a stationary tyre, the tyre will be deformed at the bottom. As the tyre contact patch is distorted as shown in figure 2. 6. 2. 1. There is a difference in direction between wheel and tyre called the slip angle. At the end of the mechanical trail a self-aligning torque is produced, the resultant force is known as the cornering force (Clancy, S., 2006).

Carroll (2002) points out that increasing the lateral force will increase slip leading to the forward movement of the breakaway point at the trailing edge towards the contact patch centre. This mechanism progresses, determined by the coefficient of friction, until the entire print begins to slide at the breakaway condition.

2. 6. 2. 2 Load Sensitivity

In metal-on-metal friction the coefficient of friction is constant over an arbitrary load range, the same is not true for tyres. With tyres the relationship is non-linear as with increasing load the ability of the tyre to produce lateral load is reduced. This is called load sensitivity (Carroll, W., 2002).

2. 6. 2. 3 Longitudinal Force, Longitudinal Slip

Figure 2. 6. 2. 3. 1: tyre print characteristics - driving (Milliken, W. F. & Milliken, D. L., 1995).

The applied driving torque, M_T , results in a tractive force, F_T , in the contact patch. Consequently, as demonstrated in figure 2. 6. 2. 3. 1 where C_1 is evidently smaller than C_2 , the tread elements in front of the axle line are compressed (Carroll, W., 2002). Clancy (2006) describes how the circumferential velocity drops as the radius of the tyre is reduced in the contact patch. Analysing relative motion of the tyre and road, it can be seen that there is no slip in the first third of the contact patch but as the treads pass through the final two thirds, the circumferential speed increases and the tread elements are stretched and backward slip occurs.

Figure 2. 6. 2. 3. 2: tyre print characteristics - braking (Milliken, W. F. & Milliken, D. L., 1995).

In the case of braking the applied braking torque acts counter to the previously described driving torque, this produces a braking force in the opposite direction to wheel travel. This force causes the print to be moved backward, stretching the tyre as demonstrated at C_3 in figure 2. 6. 2. 3. 2, and compressing the tread aft of the patch. Clancy (2006) explains that the total longitudinal braking force consists of contributions from both free rolling and the superimposed additional shear force caused by braking. Again slip velocities develop in the final two thirds of the contact patch, but in the forward direction.

3 Model Building

Having explored the different components and the theoretical performance of the suspension system, its performance must now be investigated. In order to achieve this a model is developed in Adams/Car. Where

performance is not found to be satisfactory, appropriate modifications will be made to the geometry or component selection, in line with concepts in section two. Having made any changes, the model must be re-examined and the results compared to the initial model. This process is repeated iteratively until desirable results are achieved.

3. 1 Model Subsystems

3. 1. 1 Front Suspension

Figure 3. 1. 1: Adams/Car model of front suspension assembly

The wheel connects to the upright (shown blue in figure 3. 1. 1) through a revolute joint hub bearing. The two upper and two lower arms (grey) are connected at the uprights using spherical joints and to the car using revolute joints. The tie rod (light yellow) is also connected to the upright via spherical joint, this will connect to the steering rack through another spherical joint. The rockers (yellow) connect to the pull-rods (light blue) and dampers with Hooke joints, and to the chassis through revolute joints. The pull-rods connect to the uprights with spherical joints. The communicators act between the front suspension and the body, steering, front anti-roll bars and front wheels.

3. 1. 2 Rear Suspension

Figure 3. 1. 2: Adams/Car model of rear suspension assembly

The wheel hubs, uprights, chassis, control arms, push rods, rockers and dampers are connected using revolute, spherical and Hooke joints in the same manner as the front suspension. The drive shafts (dark grey in figure 3.

1. 2) connect to the spindles and chassis via convel joints and the spindles connect to the uprights using revolute joints. The toe linkages connect to the uprights and lower control arms via spherical and Hooke joints respectively. The communicators act between the rear suspension and chassis, steering, rear anti-roll bars and rear wheels.

3. 1. 3 Steering

The steering is a rack and pinion system as modelled in figure 3. 1. 3. The steering rack is connected to the tie rods via spherical joints. The rack is connected to the rack housing (red) with a transitional joint and to the steering shaft via revolute joint. The communicators act between the steering system and the front suspension and chassis.

3. 1. 4 Wheels

To develop the wheel model tyre data is needed from the manufacturer. Fortunately, companies like Hoosier Racing Tires provide some spring rate and deflection data fo